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THE EFFECT OF VIBRATION ON HEAT TRANSFER BY FORCED CONVECTION IN A HORIZONTAL TUBE

WARREN ARTHUR GROSSETTA
AND
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THE EFFECT OF VIBRATION ON HEAT TRANSFER BY FORCED CONVECTION IN A HORIZONTAL TUBE

-

W. A. Grossetta

R. M. George

THE EFFECT OF VIBRATION ON HEAT TRANSFER
BY FORCED CONVECTION IN A HORIZONTAL TUBE

by

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Submitted in partial fulfillment
of the requirements
for the degree of
MASTER OF SCIENCE
in
MECHANICAL ENGINEERING

United States Naval Postgraduate School
Monterey, California
1953

This work is accepted as fulfilling

the thesis requirements for the degree of

MASTER OF SCIENCE
in
MECHANICAL ENGINEERING

from the

United States Naval Postgraduate School.

PREFACE

It has been noted that, on occasion, certain standard industrial heat exchanger equipment, when used in the presence of operating machinery and consequently subjected to sympathetic vibrations, performed beyond expectations and was labeled overdesigned. It seems reasonable to hypothesize that vibration has increased fluid turbulence near the surface, a primary factor in the transfer of heat by convection. A partial investigation of the phenomenon was made under conditions of free convection by R. C. Martinelli and L. M. K. Boelter (2). The object of the present work was to determine the effect of vibration, as regards both amplitude and frequency, on heat transfer to water by forced convection in a horizontal tube.

The authors wish to express appreciation to Professor E. E. Drucker for his helpful guidance.

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TABLE I

SYMBOLS AND ABBREVIATIONS

A	Area of heat transfer surface. (ft^2)
a	Amplitude of vibration. (in)
c	Specific heat of fluid at bulk temperature. (BTU/lb °F)
D	Inside diameter of tube. (ft)
Δt	Temperature difference between surface and bulk fluid. (°F)
f	Frequency of vibration. (cps)
G	Mass velocity of fluid. (lb/hr ft^2)
h	Surface coefficient of heat transfer. (BTU/hr ft^2 °F)
k	Thermal conductivity of fluid at bulk temperature. (BTU/hr ft °F)
L	Length of heated tube. (ft)
$\dot{m}$	Mass rate of fluid flow. (lb/hr)
Nu	Nusselt number. (hD/k)
Pr	Prandtl number. (cu/k)
q	Rate of heat flow. (BTU/hr)
r_i	Inside radius of tube. (ft)
r_o	Outside radius of tube. (ft)
Re	Reynolds number. (DG/u)
t	Temperature. (°F)
u	Absolute viscosity of fluid at bulk temperature. (lb/hr ft)

SUMMARY

In order to determine the effect of vibration on heat transfer by forced convection in a horizontal tube, a test section of approximately one-half inch inside diameter was designed to carry water. The test section was mounted on a fatigue vibrator and subjected to controlled transverse vibrations. Standard thermocouple procedure was used for temperature determinations.

The investigation proceeded in the laminar, transitional and slightly turbulent flow regions and almost entirely in the non-boiling range. A distinct but unexpectedly adverse effect of vibration on the surface heat transfer coefficient was noted at the lower flow rates, decreasing to a negligible effect for turbulent flow. It was not until boiling conditions were approached in the laminar and transitional regions, with either vapor or undissolved gases probably existing as compressible bubbles in the stream, that a trend towards betterment of heat transfer was found. The possibility was indicated that compressibility is a necessary condition for increased heat transfer with vibration.

DESIGN CONSIDERATIONS

As is frequently the case in experimental work one of the first problems was to design and build a test section. In accomplishing this it was necessary to answer the following questions:

1. Should the tube be thin-walled or thick-walled?
2. What material should be used for the tube?
3. How could maximum heat be best supplied?
4. How could the heat losses be minimized?

A thick-walled cylinder was selected in order that the distortion of the heat flow pattern caused by the thermocouple wells would be a minimum, for ease of placement of thermocouples, and for rigidity of the test section. The material chosen for the cylinder or tube was copper because of its high thermal conductivity and resulting small temperature drop across the tube. Then by having the bead of the thermocouple in a known position near the inner surface of the pipe it was permissible to use a straight line extrapolation to obtain the inner surface temperature with negligible error.

In order to obtain the most even distribution of heat along the test section it was decided to wrap the pipe with electrical resistance wire. It was concluded that #17 Nichrome V would give the maximum possible heat input without exceeding the temperature limits of the coil. This conclusion was based on the availability of a 220 volt power supply and on the assumption that a diameter of wire plus insulation would permit ten turns per linear inch of test pipe.

The following steps were taken to reduce the heat losses to a value which could be considered negligible:

1. A 1/8 in. deep groove was cut at each end of the cylinder.

2. The ends of the tube were further insulated with bakelite.

3. The heating coil was covered with several layers of insulation.

4. Hot air at a temperature equal to that of the outer surface of the insulation was circulated in the air space provided, effectively insulating the test section from radial heat loss.

ASSEMBLY OF THE TEST SECTION

A cold-drawn seamless copper tube with an inside diameter of 0.524 in. and a wall thickness of 0.263 in. was made into the test cylinder of overall length 12 3/8 in. Grooves 1/16 in. by 1/8 in. were cut around the pipe near each end making the actual heated length 12 in. The purpose of these grooves was to impede the axial flow of heat through the ends of the pipe.

Four small radial thermocouple wells were drilled using a #56 drill and finished with a flat bottom using a #56 flat-end drill. The spacing of these wells is shown in Figure IX. Thermocouples were installed following the procedure recommended by H. Dean Baker (1) for measurement of temperature in solids. It was found, however, that satisfactory thermocouple beads could be welded using a small oxygen-natural gas flame and a proper flux in lieu of the condenser discharge procedure. Once the wall thermocouples were cemented, the pipe was placed between two bakelite flanges and clamped in the vibration jig.

The end flanges were machined from 2 in. thick slabs of bakelite. Bakelite was chosen because of its insulating, heat resistant and mechanical properties. The insulating property of this material was necessary for reducing end heat losses. The joint between flange and test cylinder was made by butting the two pieces together and by using a silicone rubber O-ring gasket to make the joint water tight. Silicone rubber was selected because of its high temperature properties. This gasket was positioned by a V-groove machined in the flange. The two flanges and the tube were held in position by four tie-bolts,

two of which served as the electrical power busses.

The next step in the assembly procedure was to wrap the pipe with the Nichrome V wire, but first it was necessary to electrically insulate the heating wire from the copper pipe. Since the maximum allowable diameter of the wire plus insulation was 0.1 in., and assuming that the circulating fluid would keep the insulation temperature below 1000°F, fiber glass, which had been tested in a furnace to 1200°F, was chosen. A double layer of fiber glass insulation was placed on the #17 Nichrome V wire especially for this project by the Driver-Harris Company of Harrison, New Jersey. This insulation, however, was very fragile and even with the most careful handling it frayed considerably when wrapped. A thin coat of glyptal insulating varnish enabled the wire to be wrapped without danger of damaging the glass. The glyptal was not used as added insulation but merely as a binder during the period when the wire was being handled. In fact it could be burned off by applying voltage to the coil as soon as it was in place and before installing the covering insulation.

The resistance wire was wound in as tight a coil as possible to insure the heat contact between the pipe and wire and to give the most even supply of heat to the pipe. Approximately sixty feet of wire with a resistance of 19.5 ohms at room temperature were used. Number 14 copper wire was silver soldered to the Nichrome V wire for leading from the bus tie-bolts to the coil. Therefore little or no heat was generated except in the heater coil which was entirely on the 12 in. section of the pipe. After the coil was secured to the power leads a single layer of asbestos tape was wrapped over the heater. This helped to hold the coil in place and protect the 85% magnesia pipe insulation, which was used as the basic thermal insulation.

The 85% magnesia was ordinary steam pipe insulation which fitted snugly over the inner layer of asbestos and was secured in place with another wrapping of the asbestos tape (Figure I). This made the overall diameter of the insulation just under four inches and provided an annulus for an air space when the aluminum sheet outer covering was installed around the entire test section.

Four thermocouples were placed around the outer surface of the insulation for determining the temperature there. The bakelite flanges were fitted for leading hot air to the heat block. Thermocouples were cemented in these entering and leaving air passages, and compressed air warmed by a separate heating coil was led into the annulus.

Copper tube adapters were threaded into the off sides of the end flanges and the water entered and left the test section through attached flexible rubber hoses. Provision was made for determining the inlet and outlet water temperatures by installing probes in these adapters.

ASSOCIATED TEST EQUIPMENT

The associated test equipment used in this investigation consisted of Westinghouse Vibration Fatigue Equipment with an audio oscillator, the external power circuit, the hot air circuit, temperature determining apparatus and a calibrated orfice.

1. Westinghouse Vibration Fatigue Equipment.

This equipment consisted of the vibrator (Figure III), and the control panel and audio oscillator (Figure IV). The arrangement allowed vibration of the test section from 20 to 20,000 cycles per second. Fairly large amplitudes could be obtained at the low frequencies, but as frequency increased the amplitude obtainable was decreased. The amplitude was measured with a traveling microscope (Figure I). In order to use the equipment it was necessary to manufacture a support jig to connect the test section to the actuating rod of the vibrator and at the same time to prevent any tilting of the vibrator coil due to unsymmetrical loading. This frame can be seen in Figure I and consisted of

a. Two vertical support posts.

b. The cross-bar which was connected to the vibrator actuating rod by an adapter and which was prevented from tilting by two brass sleeve bearings that slid on the support posts.

c. Two vertical support clamps which held the bakelite flanges and were secured to the cross-bar.

2. External Power Circuit

The equipment comprising this circuit was standard alternating current apparatus.

a. Variac, 0-135 volts.

b. Transformer, one-to-one.

c. Ammeter, 0-5 and 0-20 amperes.

d. Voltmeter, 0-150 and 0-300 volts.

e. Wattmeter, 0-1500 and 0-5000 watts.

3. Hot Air Circuit.

The unit consisted of an air supply, a resistance wire heating coil mounted in a pyrex tube, a variac to control the voltage across the heating coil and the rubber tube leads necessary to conduct the air to the heater and from the heater to the test section annulus (Figures III and IV).

4. Thermometry Apparatus.

The equipment and arrangement used for determining the various temperatures from the thermocouples installed in the test section were those recommended by H. Dean Baker (1), using a common ice junction. Instead of the recommended Leeds and Northrup switch box, double pole knife switches were used.

5. Calibrated Orfice.

In order to determine the rate of flow of water during each run it was necessary to manufacture and calibrate an orfice (Figure VIII). The pressure drop across the orfice was measured by a mercury manometer and the flow rate ascertained from the previously determined calibration curve.

Figure [illegible]

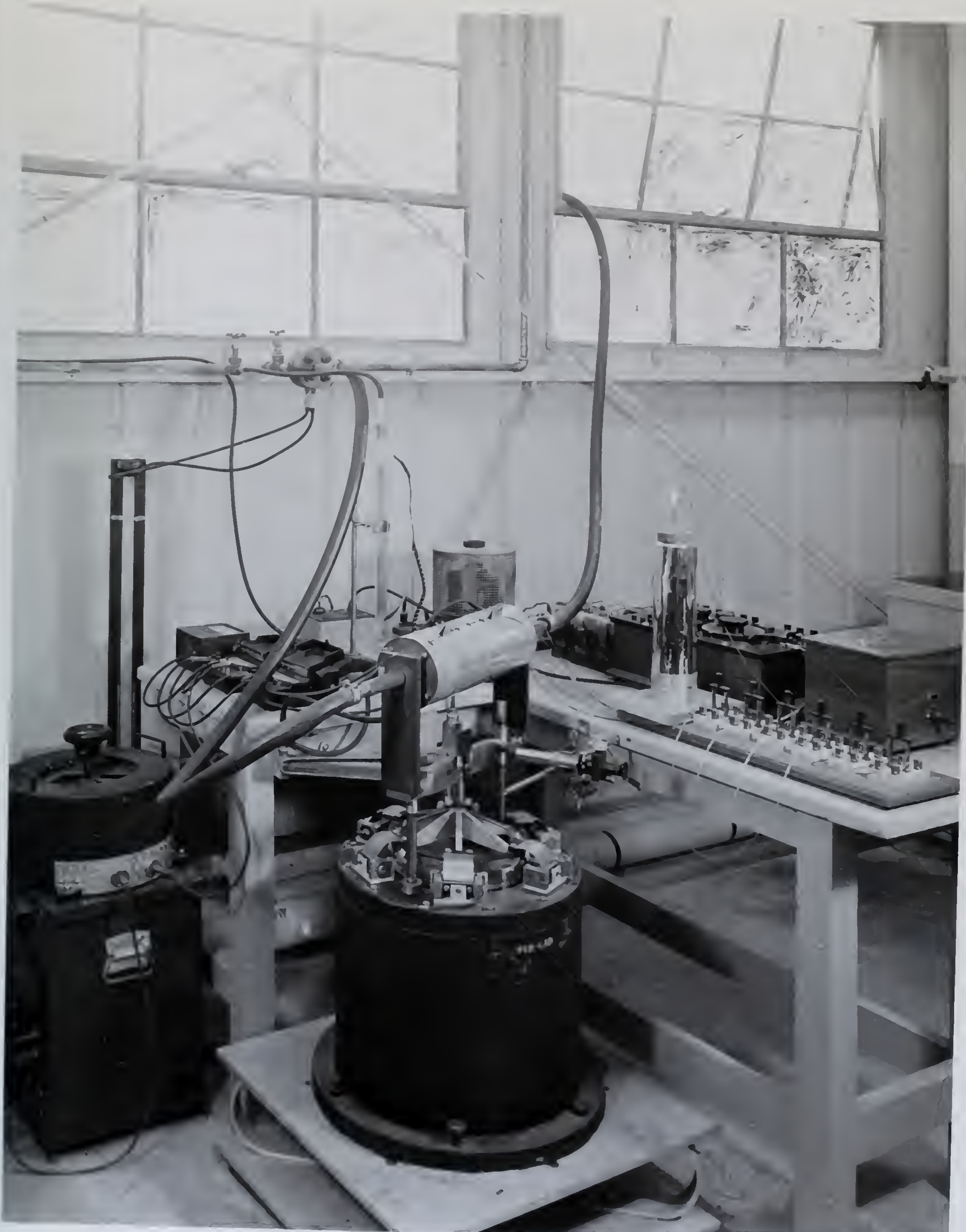

Figure II

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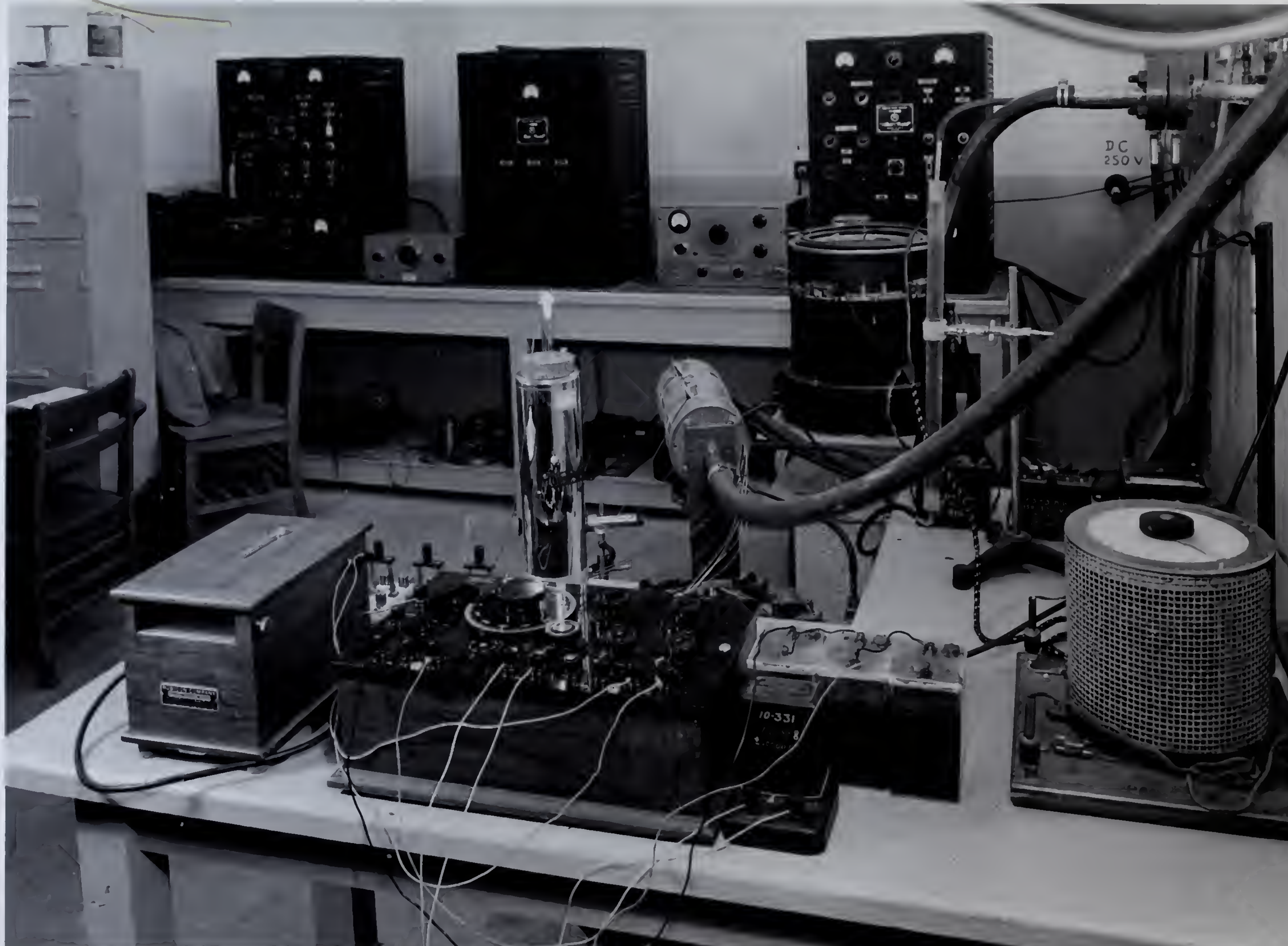

Figure IV

PROCEDURE

Prior to commencement of a run, the flow of water was adjusted to the desired rate and the proper voltage applied to the test section. The time necessary to obtain steady state conditions, including adjustment of temperature of the surrounding air to that of the insulation surface, was about one and one-half hours although it depended somewhat on the amount of change in the flow rate between runs.

On each run the readings from the four tube-wall thermocouples and from the inlet and outlet stream probes were recorded, first without vibration of the test section and then for each condition of vibration. As a starting point in the investigation two different amplitudes at two frequencies, 20 and 60 cycles per second, were arbitrarily chosen for taking data. 20 cps was the minimum frequency which the vibrator could produce. It was soon apparent that, for non-boiling, 60 cps was within the range of maximum effect on the heat transfer coefficient, and so the same four basic conditions of vibration were maintained throughout the investigation. These, along with the non-vibrating part of each run, yielded the data for the presentation of Figure V. Wider ranges of frequency and amplitude were undertaken on runs 5, 6 and 7, which produced the results shown in Figure VI.

It was found to be very difficult to reproduce an exact amplitude of vibration from one run to the next and consequently only an approximate value was achieved each time. This is considered justified in view of the minor effect of amplitude as compared to frequency.

The reduction of data and the computations involved certain assumptions. It was assumed that recorded temperatures were those at the centers of the thermocouple beads, which averaged 0.04 in. in diameter. Calculations were based on constant temperature gradients in a radial direction across the tube wall, and in axial directions through the tube wall and in the stream. The average recorded inlet water temperature for a run was used, along with a temperature rise computed from the heat input and flow rate. The heat input is believed to have been measured quite accurately in view of the precautions taken to minimize losses, while the stream probe on the outlet side is not considered to have given reliable indications due to insufficient allowance for mixing to bulk water temperature at the location of the thermocouple.

RESULTS

For a heat flux input to the test section of approximately 3400 BTU/hr (or a heat flux density of about 25000 BTU/hr ft^2), and for laminar, transitional and turbulent flow without tube vibration, the findings are presented in Figure V in the form of the following points plotted to coordinates of $Nu/Pr^{0.4}$ versus Re:

1. Values computed by the formula

   $hD/k = 0.023\ (DG/u)^{0.8}\ (cu/k)^{0.4}$ which, according to W. H. McAdams (3), gives quite good correlation with experimental data for fluids having viscosities not more than twice that of water and for Reynolds numbers exceeding 2100.

2. Values determined experimentally without vibrating the test section, where $h = q/A\Delta t$. (Δt is the arithmetical mean difference between surface and bulk water temperatures through the test section. See Sample Calculations, Appendix II).

3. Values determined experimentally by vibrating the test section at

   a. 20 cps, approximate amplitude 0.03 in.

   b. 20 cps, approximate amplitude 0.04 in.

   c. 60 cps, approximate amplitude 0.03 in.

   d. 60 cps, approximate amplitude 0.02 in.

The diminishing effect of the vibration at very low flow rates, where water temperature rise was relatively large, indicated the possibility of the reverse and expected trend towards betterment of heat transfer as compressible bubbles of vapor and previously dis-

solved air began to form in the stream. Accordingly, input power was doubled on run 15 and data was taken at a Reynolds number of about 4000, after which maximum voltage was put on the heating element, corresponding to a heat flux density of 60,300 BTU/hr ft^2, and runs were made at progressively lower Reynolds numbers. Early in this final stage two of the tube wall thermocouples were damaged. Computations therefore necessarily involved only the local temperature taken at the position of the final thermocouple in the test section.

Figure VI is simply a cross-plot of Figure V using the data of runs 5, 6 and 7 where greater ranges of vibration variables were recorded.

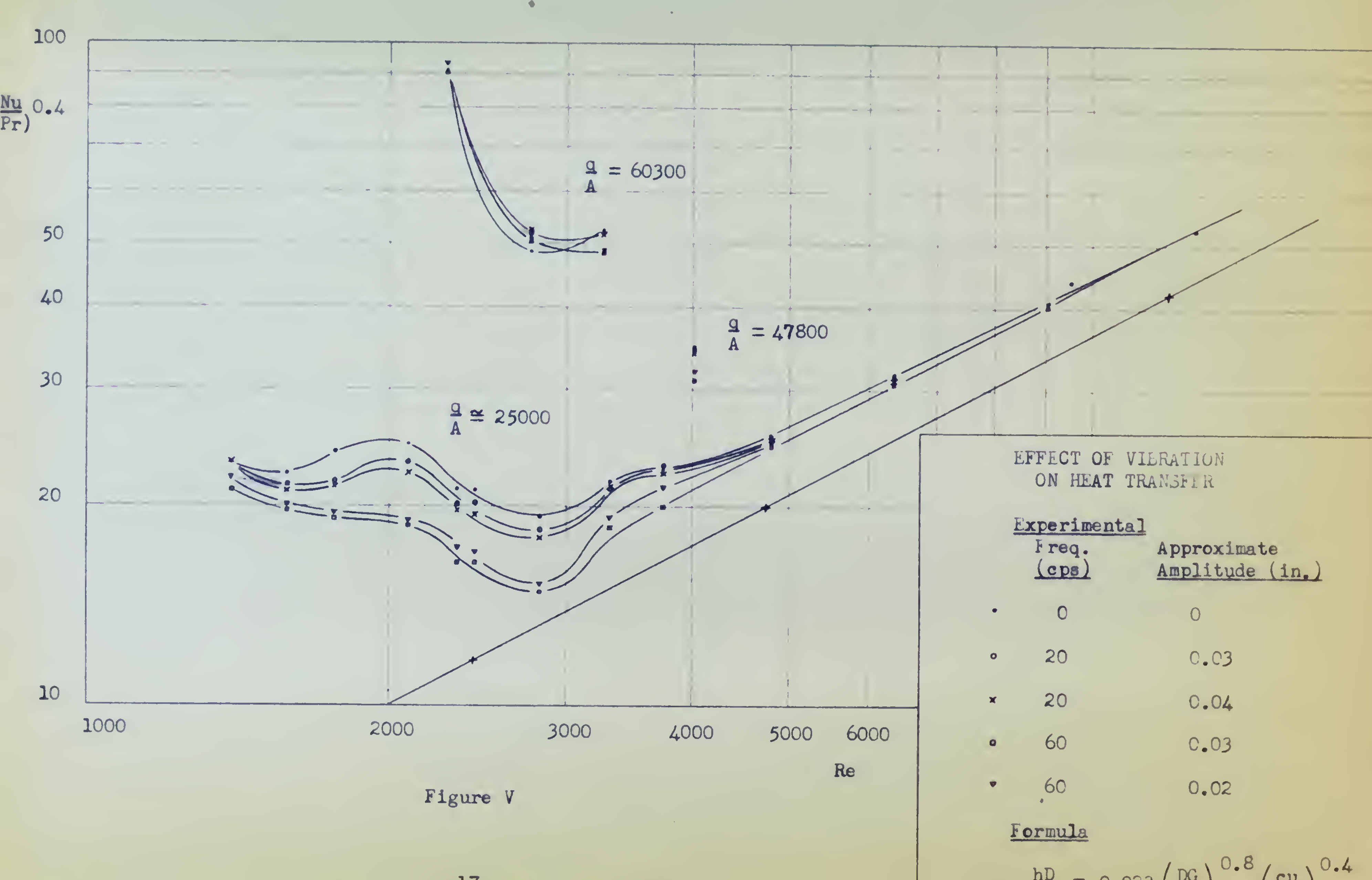
100
50
40
30
20
10
Nu
Pr) 0.4
1000
2000
3000
4000
5000
6000
Re
g/A = 60300
g/A = 47800
g/A ≅ 25000
EFFECT OF VIBRATION
ON HEAT TRANSFER
Experimental
Freq. (cps)
Approximate Amplitude (in.)
0 0
20 0.03
20 0.04
60 0.03
60 0.02
Formula
hD = 0.023 (DG)^0.8 (cu)^0.4

Figure V

CONCLUSIONS

In the non-boiling range and for laminar and transitional flow, vibration may be seen to cause an unmistakable decrease in the heat transfer coefficient which is apparently dependent on frequency, and to a lesser degree on amplitude. The effect is found to be best correlated by plotting to a parameter product of frequency and the seventh root of amplitude (Figure VI). The vibration effect is further seen to decrease with increase in Reynolds number, to become negligible at values of the number above 5000, and to disappear in the turbulent flow region.

The results of the final runs under maximum power, although not considered quantitatively as good as earlier findings for reasons given in the previous section, strongly indicate that a betterment of heat transfer with vibration may be expected when compressible bubbles of vapor and undissolved gases begin to form in the stream.

It is plausible in retrospect to accept a compressibility factor in the fluid as a necessary condition for betterment of heat transfer with vibration, in that gaseous bubbles provide space for increased turbulence of the remaining liquid. The only explanation offered for the phenomenon of adverse effect in the absence of compressibility is that a flow disturbance, possibly a swirl about the tube axis, is set up by the vibration which interferes with natural convection processes. Such interference is then obscured as Reynolds number is increased and conditions of turbulent flow begin to predominate.

The fact that the nominal value of the heat transfer coefficient

increases sharply with heat flux density is readily explained by the fact that the transition-to-boiling range is involved. It may be noted from the data that the change in temperature difference with change in heat flux density is relatively small at equal Reynolds numbers. See W. H. McAdams (4).

Thermocouple error was checked at the boiling point of water and found to be minus 2°F. Since computations employed only temperature differences, and water properties of minor variation with temperature, it is not considered necessary to have run a calibration curve.

Experimental results for Reynolds numbers above 5000 agree very closely with the generally accepted experimental data represented by W. H. McAdams (3).

RECOMMENDATIONS

It is recommended that any further investigation in the field of forced convection heat transfer under conditions of vibration be pursued with the following modifications:

1. Within the boiling range in order to determine the effect of compressibility in the stream. Towards this end the inlet water may be preheated to near boiling and a test section of greater power capacity may be designed. The test section used on this project was designed for maximum power assuming a thicker wire insulation than that actually supplied by the manufacturer. Available voltage therefore did not produce full load current for #17 Nichrome V wire and maximum possible heating was not achieved.
2. With de-gassed water and with air in order to further determine the extreme effects of both non-compressibility and compressibility in the stream.
3. With longitudinal vibration of the test section.

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[illegible]

1. [illegible]

2. [illegible]

3. [illegible]

4. [illegible]

APPENDIX I

TABLE II

RECORDED DATA

Run	f	a	$\dot{m}$	I	V	t_1	t_2	t_3	t_4	t_5	t_6
1a	0	0.0	220	7.10	140	139.8	145.8	155.7	160.3	65.3	80.5
b	20	.0281				143.8	151.3	157.4	164.2	65.6	81.4
c	20	.0458				146.0	156.5	160.0	163.0	65.6	80.9
d	60	.0335				160.2	175.0	174.3	182.8	66.1	80.6
e	60	.0200				154.8	172.3	174.9	179.0	66.2	80.3
2a	0	0.0	520	7.10	140	106.7	116.2	120.6	120.9	66.1	71.6
b	20	.0270				106.7	116.2	121.2	121.6	66.4	71.2
c	20	.0448				107.0	117.3	121.2	121.7	66.4	72.3
d	60	.0348				107.4	118.1	121.3	122.9	66.7	72.3
e	60	.0185				107.7	118.2	122.2	123.3	67.2	72.0
3a	0	0.0	157	7.10	140	133.0	134.0	139.1	142.9	65.4	87.6
b	20	.0282				135.0	139.1	141.9	148.7	65.8	89.0
c	20	.0446				137.4	142.2	145.9	149.1	66.0	90.0
d	60	.0350				152.5	157.5	157.5	161.5	66.0	89.0
e	60	.0195				149.4	157.9	155.1	160.5	66.0	90.6
4a	0	0.0	301	7.10	140	129.4	131.7	137.4	145.3	65.2	75.4
b	20	.0281				124.6	133.3	138.5	142.5	65.4	75.4
c	20	.0448				127.3	134.7	139.8	145.6	65.5	75.8
d	60	.0340				134.7	147.0	146.0	148.7	65.4	74.7
e	60	.0195				133.3	137.0	142.2	148.4	65.2	75.0
5a	0	0.0	257	7.10	140	134.0	138.1	145.0	153.1	67.4	81.0
b	20	.0273				132.3	140.8	147.3	154.8	67.4	80.5
c	20	.0446				131.3	140.8	148.4	154.2	67.4	80.7
d	60	.0307				138.4	152.4	160.5	164.2	67.4	80.0
e	60	.0193				137.0	150.4	158.2	163.5	67.3	80.0
f	25	.0194				136.7	145.3	155.2	158.6	67.3	80.6
g	25	.0451				136.7	150.1	156.0	161.3	67.2	80.5
h	30	.0182				139.8	148.7	154.4	160.5	67.2	81.0
i	30	.0421				137.4	151.4	156.4	161.2	67.2	80.9
j	100	.0078				136.7	151.9	158.3	163.7	67.4	81.1
k	200	.0035				134.6	152.8	160.1	165.2	67.4	80.8
l	300	.0032				140.3	154.0	159.6	166.0	67.5	81.0
m	500	.0031				139.4	152.7	158.3	160.5	67.5	80.5

TABLE II

RECORDED DATA

Run	f	a	$\dot{m}$	I	V	t_1	t_2	t_3	t_4	t_5	t_6
6a	0	0.0	180	7.10	140	140.1	142.9	153.1	159.2	67.0	86.1
b	20	.0257				142.5	147.7	154.8	162.2	67.4	87.2
c	20	.0451				143.5	150.7	159.2	167.6	67.7	87.2
d	60	.0328				148.0	168.5	178.0	181.1	68.2	87.2
e	60	.0186				149.4	163.9	172.6	178.0	68.3	88.4
f	25	.0198				151.8	153.2	158.5	166.4	68.1	88.2
g	25	.0390				151.8	159.9	166.0	173.6	68.0	87.3
h	30	.0185				154.1	163.5	170.1	172.8	68.0	86.5
i	30	.0412				157.4	166.5	173.0	174.7	68.1	87.4
j	100	.0090				151.6	165.9	177.3	181.4	68.0	88.0
k	200	.0027				154.1	169.2	171.0	173.5	68.0	87.9
l	300	.0031				153.8	167.1	176.6	182.2	67.8	87.4
m	500	.0032				153.1	165.2	174.2	179.5	67.7	87.5
7a	0	0.0	387	7.10	140	114.4	126.9	134.2	139.3	66.0	72.8
b	20	.0279				114.6	128.6	136.2	138.3	66.1	72.8
c	20	.0447				113.8	127.5	135.9	139.5	66.8	73.8
d	60	.0331				115.4	130.2	137.0	139.7	67.0	74.0
e	60	.0199				115.2	129.1	136.9	139.8	67.0	73.7
f	30	.0181				114.9	128.2	136.0	139.7	67.0	73.2
g	30	.0419				115.5	128.5	136.1	140.1	66.9	73.1
h	100	.0028				115.5	128.6	135.2	140.6	66.9	75.0
i	300	.0028				115.4	128.3	136.8	140.6	66.9	74.2
j	500	.0031				116.2	128.8	137.4	141.0	66.9	73.8
8a	0	0.0	763	7.10	140	97.0	104.1	106.0	107.5	65.8	68.7
b	20	.0287				97.0	104.4	106.0	107.6	65.8	68.7
c	20	.0444				97.4	104.5	106.8	108.2	65.6	68.9
d	60	.0328				97.1	104.5	107.0	108.4	65.5	68.9
e	60	.0195				97.0	104.1	106.6	107.7	65.8	69.1
9a	0	0.0	820	7.20	140	94.3	100.1	101.6	102.0	64.0	67.5
b	20										
c	20					No measurable change with vibration					
d	60										
e	60										
10a	0	0.0	1090	7.20	140	89.9	94.1	95.6	95.5	64.2	65.7
b	20										
c	20					No measurable change with vibration					
d	60										
e	60										

TABLE II

RECORDED DATA

Run	f	a	$\dot{m}$	I	V	t_1	t_2	t_3	t_4	t_5	t_6
11a	0	0.0	120	7.20	140	144.0	146.4	152.1	155.6	69.1	100.8
b	20	.0274				157.7	156.2	155.4	156.9	69.4	104.6
c	20	.0443				157.8	157.9	157.2	160.0	70.0	105.2
d	60	.0333				164.6	166.9	166.4	167.6	70.4	106.9
e	60	.0191				164.7	165.2	164.2	167.2	70.4	108.5
12a	0	0.0	100	7.20	140	158.9	159.2	162.2	168.5	72.5	114.0
b	20	.0302				162.5	162.9	164.2	169.8	72.9	116.3
c	20	.0459				163.4	164.8	165.8	171.5	73.0	116.2
d	60	.0315				166.4	170.0	173.0	178.0	73.1	117.0
e	60	.0199				166.1	168.6	170.3	176.9	73.3	117.0
13a	0	0.0	65	7.20	140	166.2	168.4	175.0	183.3	75.0	133.0
b	20	.0280				164.5	166.6	176.5	185.5	75.2	137.8
c	20	.0447				163.9	167.4	176.2	185.0	75.3	135.9
d	60	.0341				169.7	176.5	184.0	192.4	74.9	135.6
e	60	.0192				167.1	174.5	179.5	188.4	74.5	136.2
14a	0	0.0	172	7.20	140	141.1	149.9	153.0	157.1	67.5	90.5
b	20	.0286				141.7	152.7	157.1	163.3	67.8	92.0
c	20	.0446				141.8	157.5	159.6	164.2	68.0	92.8
d	60	.0353				160.3	174.2	174.8	174.2	68.1	95.8
e	60	.0195				159.3	168.5	168.6	170.0	68.4	96.1
15a	0	0.0	301	9.85	195	150.0	153.8	165.2	169.5	65.0	85.8
b	20	.0281				147.3	154.8	166.5	169.9	65.0	85.5
c	20	.0437				149.4	156.2	166.2	171.3	65.0	85.5
d	60	.0334				161.5	165.5	171.3	179.7	65.1	84.7
e	60	.0187				155.1	164.5	170.3	179.0	65.1	85.0
16a	0	0.0	164	11.00	220			187.1	193.5	68.6	123.5
b	20	.0299						187.3	194.5	68.7	123.4
c	20	.0459						188.2	194.8	68.7	123.9
d	60	.0360						194.2	199.8	68.3	123.1
e	60	.0196						193.6	199.3	68.2	123.3
17a	0	0.0	120	11.00	220			213.5	219.7	69.0	146.0
b	20	.0278						210.9	214.2	68.7	147.0
c	20	.0450						207.6	213.7	68.6	149.6
d	60	.0336						209.2	217.8	68.6	150.0
e	60	.0195						208.6	216.7	68.8	150.8
18a	0	0.0	65	11.00	220			231.9	237.7	70.8	211.9
b	20	.0280						233.4	238.4	71.1	213.5
c	20	.0439						230.0	238.1	71.3	213.3
d	60	.0327						234.3	237.7	71.4	213.2
e	60	.0199						234.0	236.6	71.1	211.5

APPENDIX I

TABLE III

REDUCED DATA

Run	Δt	h	k	u	Nu	$Pr^{0.4}$	$Nu/Pr^{0.4}$	Re
1a	76.4	325	0.345	2.26	41.2	2.12	19.4	2840
b	80.2	309			39.2		18.5	
c	82.4	301			38.1		18.0	
d	99.1	250			31.7		14.9	
e	96.4	257			32.5		15.3	
2a	45.9	540	0.344	2.37	68.6	2.16	31.7	6400
b	46.3	536			68.1		31.5	
c	46.8	530			67.3		31.1	
d	47.4	523			66.4		30.7	
e	47.8	519			65.9		30.5	
3a	60.2	412	0.347	2.17	51.9	2.08	24.9	2110
b	64.0	388			48.8		23.4	
c	66.7	372			46.8		22.5	
d	80.1	310			39.0		18.7	
e	78.7	315			39.7		19.1	
4a	64.4	385	0.344	2.33	48.9	2.15	22.8	3770
b	63.1	393			49.9		23.2	
c	65.3	380			48.3		22.5	
d	73.5	338			42.9		20.0	
e	68.6	362			46.0		21.4	
5a	67.6	367	0.346	2.25	46.3	2.12	21.8	3330
b	68.9	360			45.5		21.4	
c	68.8	361			45.6		21.5	
d	78.9	314			39.7		18.7	
e	76.6	324			40.9		19.3	
f	74.5	333			42.1		19.9	
g	76.7	323			40.8		19.2	
h	76.9	323			40.8		19.2	
i	77.2	321			40.5		19.1	
j	77.1	322			40.6		19.2	
k	78.8	315			39.8		18.8	
l	80.5	308			38.9		18.4	
m	78.1	318			40.2		19.0	

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TABLE III

REDUCED DATA

Run	Δt	h	k	u	Nu	$Pr^{0.4}$	$Nu/Pr^{0.4}$	Re
6a	71.3	348	0.347	2.15	43.8	2.07	21.2	2450
b	74.2	334			42.1		20.3	
c	77.6	320			40.3		19.5	
d	91.4	271			34.1		16.5	
e	88.2	281			35.4		17.1	
f	80.4	308			38.8		18.7	
g	85.3	291			36.7		17.7	
h	87.7	283			35.6		17.2	
i	90.4	274			34.5		16.7	
j	91.5	271			34.1		16.5	
k	89.5	277			34.9		16.9	
l	92.4	268			33.8		16.3	
m	90.5	274			34.5		16.7	
7a	57.2	434	0.344	2.34	55.1	2.15	25.6	4830
b	57.9	428			54.4		25.3	
c	57.9	428			54.4		25.3	
d	59.1	420			53.3		24.8	
e	58.8	422			53.6		24.9	
f	58.3	425			54.0		25.1	
g	58.7	423			53.7		25.0	
h	58.5	424			53.8		25.0	
i	58.8	422			53.6		24.9	
j	59.6	417			53.0		24.6	
8a	35.3	703	0.343	2.45	89.6	2.20	40.7	9100
b	35.3	703			89.6		40.7	
c	35.7	695			88.6		40.3	
d	35.8	693			88.3		40.1	
e	35.3	703			89.6		40.7	
9a	32.9	763	0.342	2.49	97.5	2.21	44.1	9620
b								
c			No change with vibration					
d								
e								
10a	27.5	913	0.342	2.49	116.7	2.21	52.8	12800
b								
c			No change with vibration					
d								
e								

TABLE III

REDUCED DATA

Run	Δt	h	k	u	Nu	$Pr^{0.4}$	$Nu/Pr^{0.4}$	Re
11a	64.8	387	0.350	1.975	48.3	2.00	24.2	1775
b	71.8	350			43.7		21.9	
c	73.5	342			42.7		21.4	
d	81.6	308			38.4		19.2	
e	80.5	312			39.0		19.5	
12a	71.4	352	0.353	1.843	43.6	1.938	22.5	1585
b	74.3	338			41.8		21.6	
c	75.7	332			41.1		21.2	
d	81.1	310			38.3		19.8	
e	79.8	315			39.0		20.1	
13a	71.4	352	0.359	1.630	42.8	1.832	23.4	1165
b	71.4	352			42.8		23.4	
c	71.1	353			43.0		23.5	
d	78.7	319			38.8		21.2	
e	75.5	333			40.5		22.1	
14a	71.8	350	0.347	2.14	44.1	2.07	21.3	2350
b	75.3	333			41.9		20.2	
c	77.2	325			40.9		19.8	
d	92.4	272			34.2		16.5	
e	87.8	286			36.0		17.4	
15a	82.7	578	0.346	2.19	73.0	2.09	34.9	4010
b	82.8	577			72.8		34.8	
c	83.8	570			72.0		34.4	
d	92.6	516			65.1		31.1	
e	90.3	529			66.8		32.0	
16a	79.6	758	0.364	1.453	91.1	1.742	52.3	3290
b	80.6	749			90.0		51.7	
c	80.9	746			89.6		51.4	
d	85.9	703			84.5		48.5	
e	85.4	707			84.9		48.7	
17a	89.4	675	0.371	1.265	79.5	1.634	48.6	2770
b	83.9	719			84.7		51.8	
c	83.4	723			85.2		52.1	
d	86.7	696			82.0		50.1	
e	86.4	698			82.2		50.3	
18a	54.1	1115	0.397	0.828	122.6	1.342	91.4	2290
b	54.8	1100			121.0		90.3	
c	54.5	1107			121.7		90.7	
d	54.1	1115			122.6		91.4	
e	53.0	1138			125.2		93.3	

APPENDIX II

SAMPLE CALCULATIONS

Run 3a (non-vibrating condition)

Extrapolation from tube-wall thermocouple temperatures to inner surface temperature.

$$q = \frac{2\pi kL \,\Delta t''}{\ln (r_o/r_i)}$$

$$\frac{7.10 \times 140}{0.2931} = \frac{2\pi \times 221 \times 1 \times \Delta t''}{\ln (0.525/0.262)}$$

$$\Delta t'' = 1.7\ ^\circ F \quad \text{(temperature drop across tube wall)}$$

$$\Delta t' = \frac{0.083}{0.263} \times 1.7 = 0.5\ ^\circ F \quad \text{(temperature drop from thermocouples to inner surface)}$$

Determination of arithmetical mean difference between inner surface and bulk water temperatures. (See Figure VII.)

$$\Delta t = (137.3 - 0.5) - 76.6 = 60.2\ ^\circ F$$

Calculation of average surface heat transfer coefficient in the test section.

$$h = q/A\Delta t$$

$$= \frac{7.10 \times 140/0.2931}{\pi \times 0.524/12 \times 1 \times 60.2} = 412 \text{ BTU/hr ft}^2\,{}^\circ F$$

Properties of water at average bulk temperature of 76.6 °F.

$$k = 0.347 \text{ BTU/hr ft °F}$$

$$u = 2.17 \text{ lb/hr ft}$$

Dimensionless parameters.

$$Nu = hD/k$$

$$= \frac{412 \times 0.524/12}{0.347} = 51.9$$

$$(Pr)^{0.4} = (cu/k)^{0.4}$$

$$= (1 \times 2.17/0.347)^{0.4} = 2.08$$

$$Nu/(Pr)^{0.4} = 51.9/2.08 = 24.9$$

$$Re = DG/u$$

$$= \frac{0.524/12 \times \dfrac{157}{\pi/4 \times (0.524/12)^2}}{2.17} = 2110$$

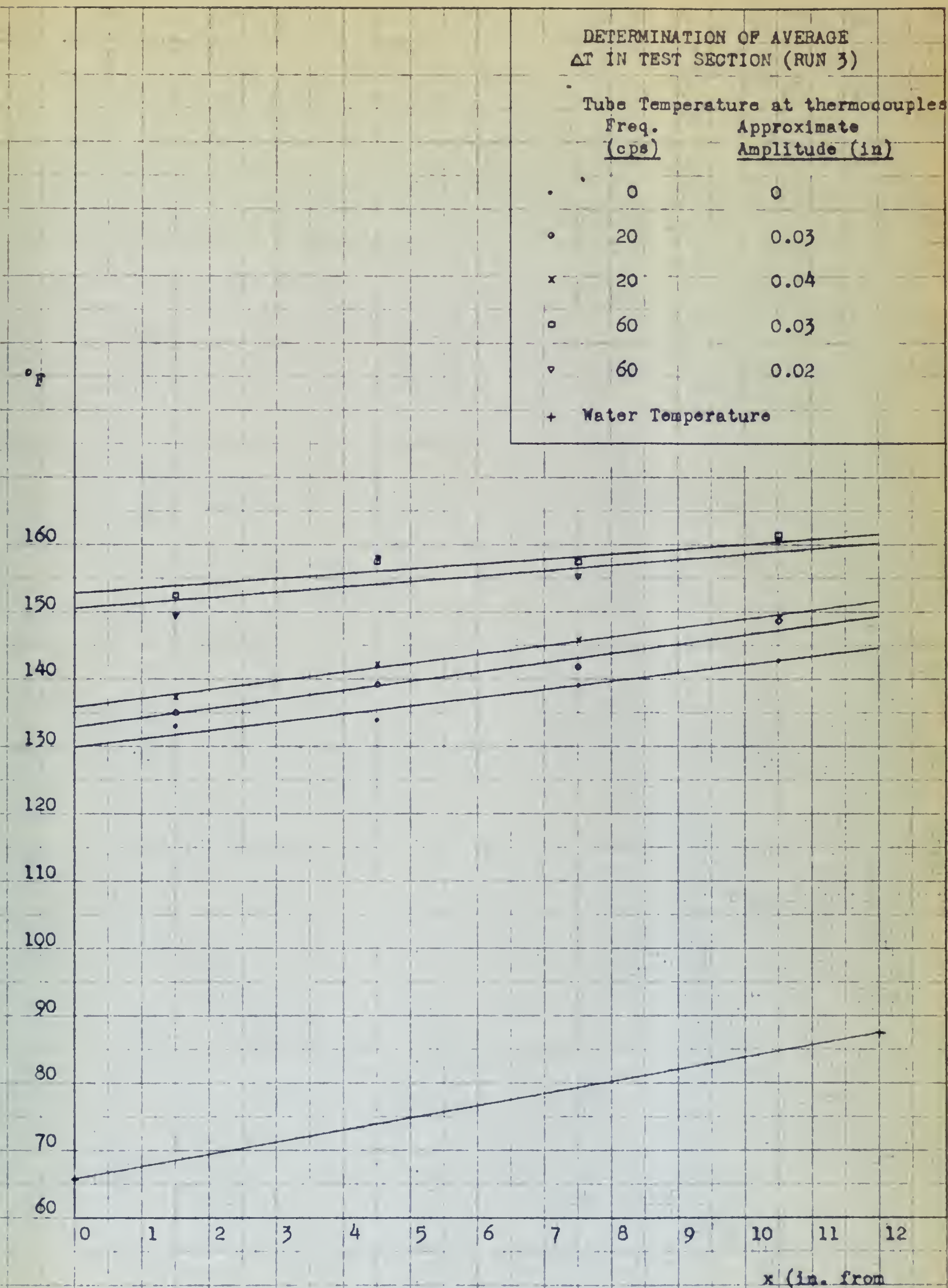
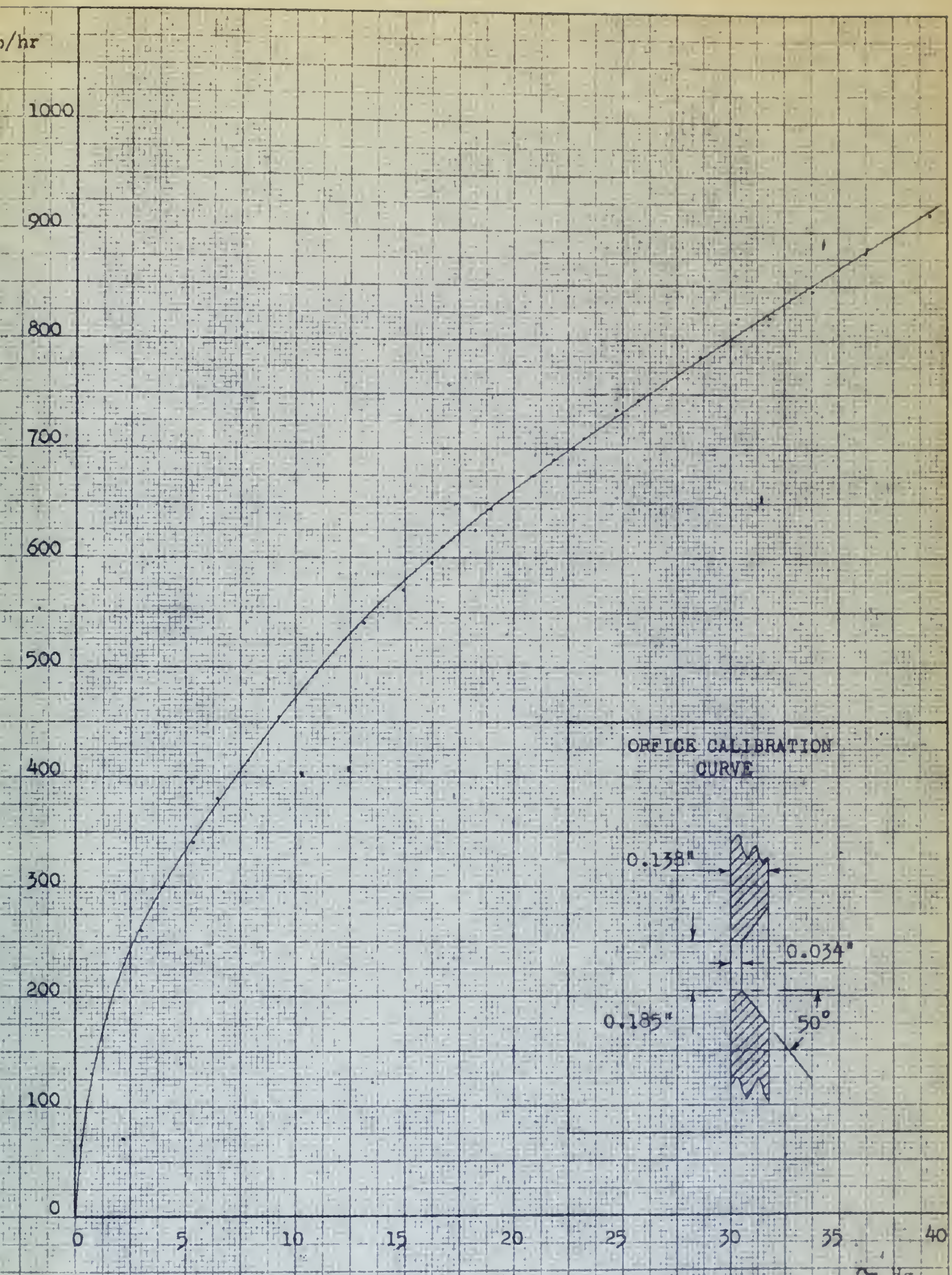
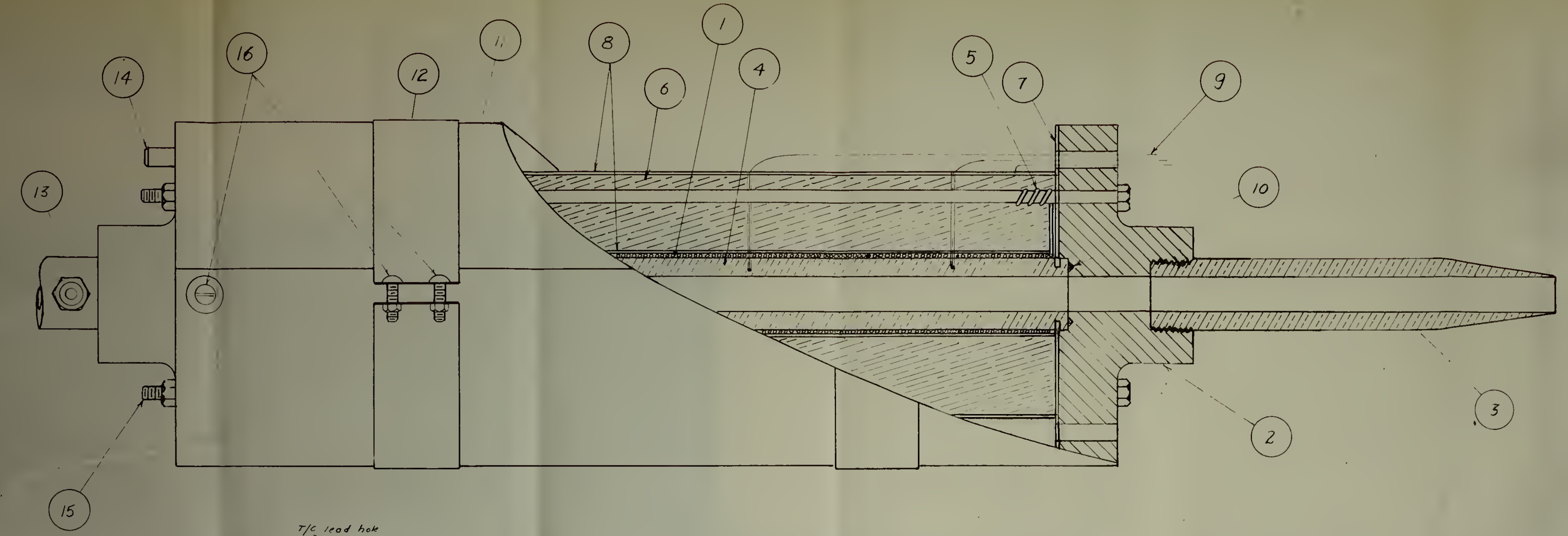

DETERMINATION OF AVERAGE
ΔT IN TEST SECTION (RUN 3)
Tube Temperature at thermocouples
Freq. (cps)
Approximate Amplitude (in)
0
0
20
0.03
20
0.04
60
0.03
60
0.02
Water Temperature
°F
160
150
140
130
120
110
100
90
80
70
60
0
1
2
3
4
5
6
7
8
9
10
11
12
x (in. from tube inlet)

Figure VII

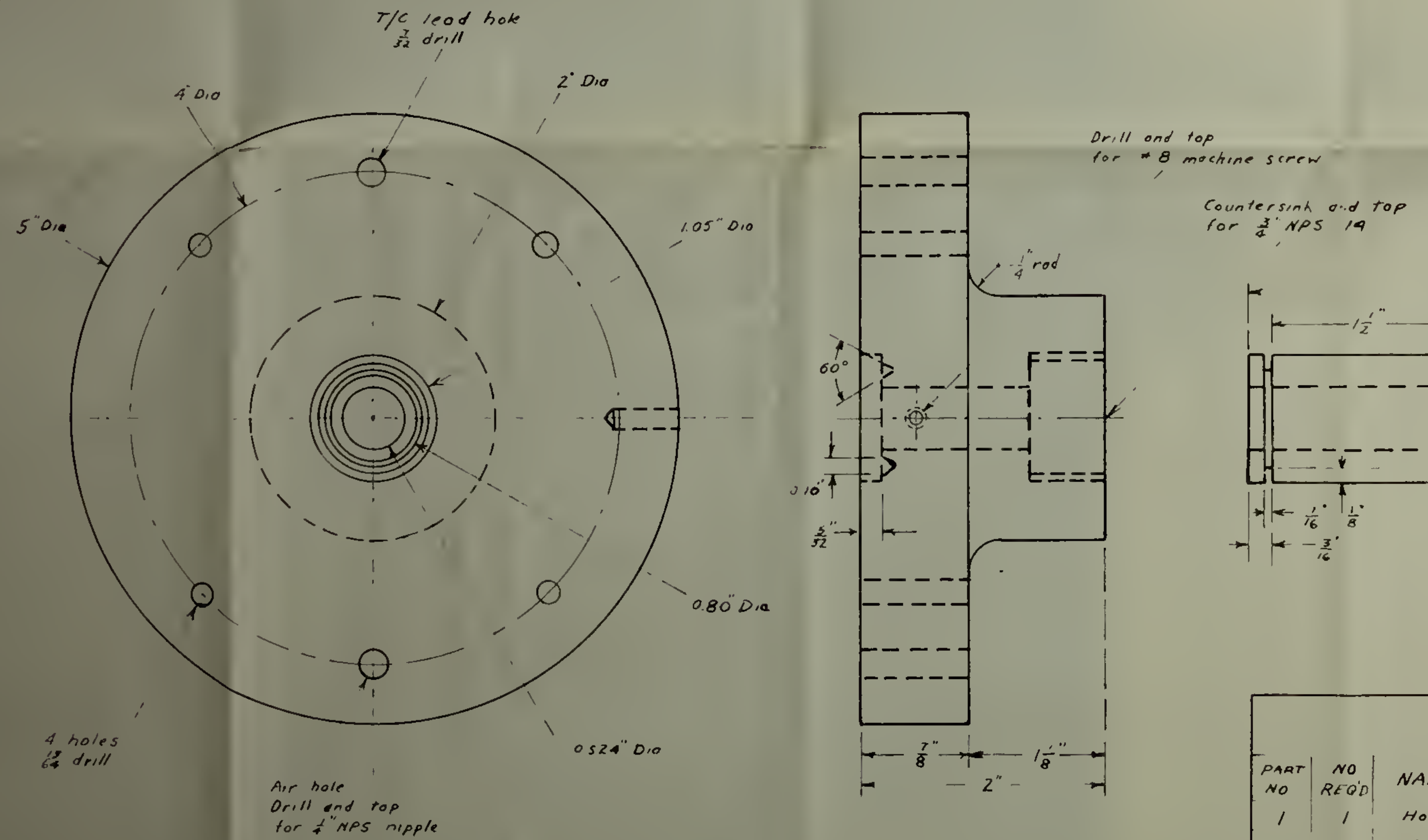


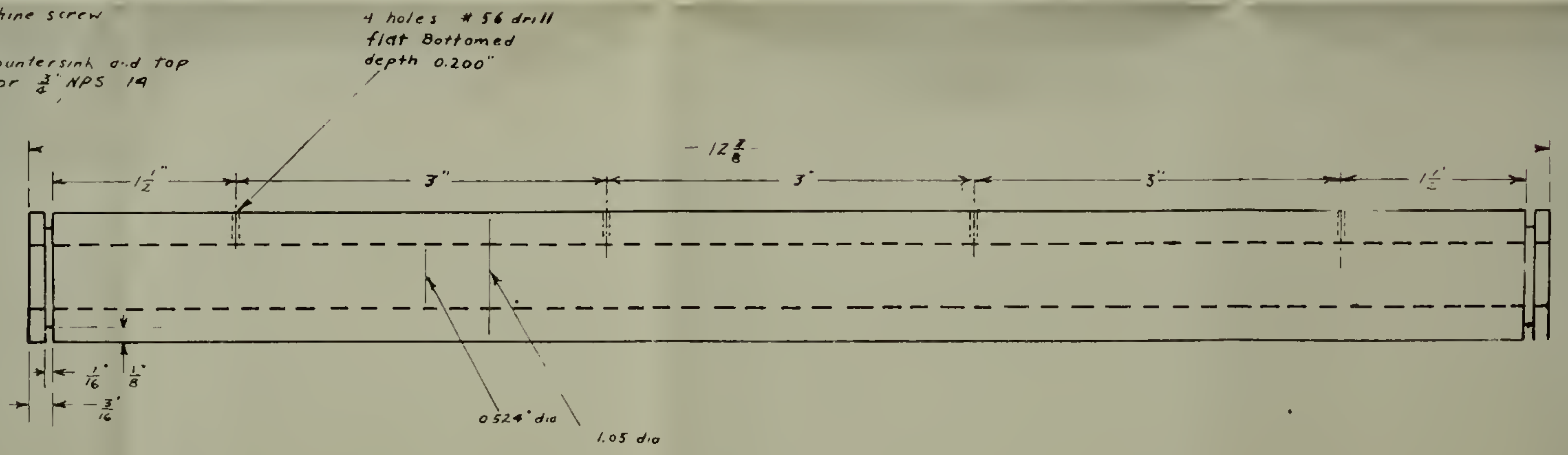
b/hr
1000
900
800
700
600
500
400
300
200
100
0
0
5
10
15
20
25
30
35
40
Cm Hg
ORFICE CALIBRATION CURVE
0.138"
0.034"
0.185"
50°

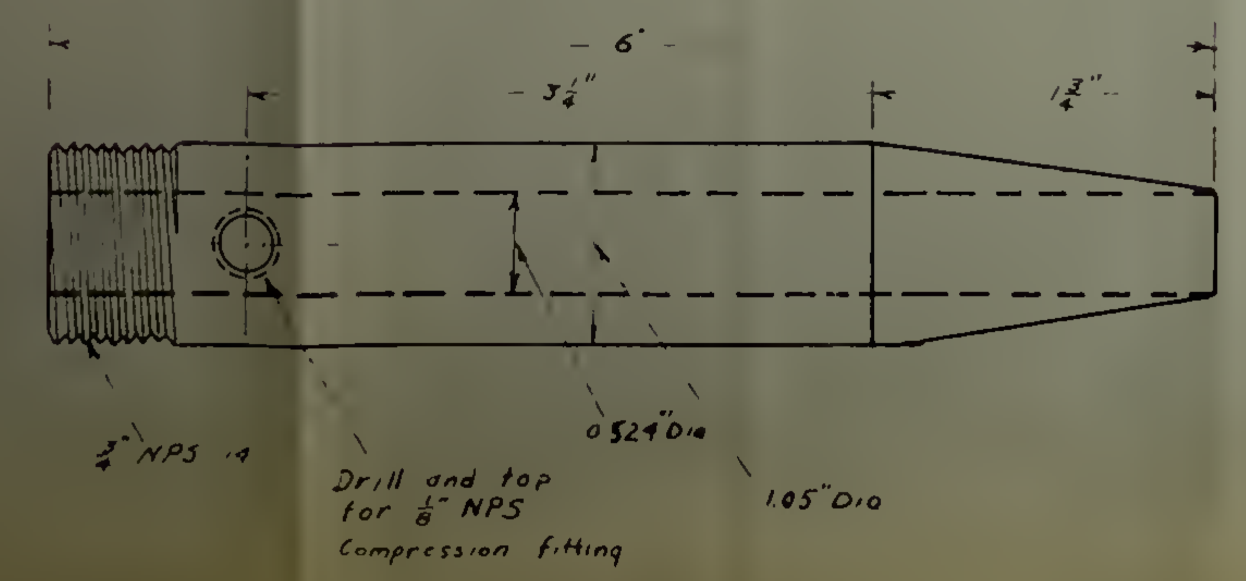
Figure VIII

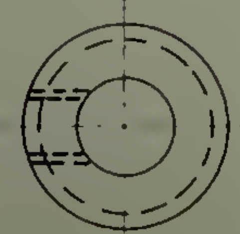


Note. Silver solder all Nichrome wire Connections









BILL OF MATERIALS

PART NO	NO REQ'D	NAME OF PART	SPECIFICATION
1	1	Hot Pipe	Copper tube, $\frac{3}{4}$" double extra heavy, seamless, cold-drawn
2	2	Flange	Bakelite
3	2	End Pipe	Same as part no. 1
4	1	Heating Element	#17 glass fibre insulated Nichrome V wire
5	2	Wire	#14 copper wire
6	1	Insulation	85 % magnesia
7	2	Insulation	$\frac{1}{32}$" asbestos sheet
8	1	Tape	1" asbestos cloth tape roll
9	12	Thermocouple	#30 Iron-constanton duplex
10	2	O-Ring	$\frac{11}{16}$ I.D silicone rubber
11	1	Cover	Aluminum Sheet 14 x 16
12	2	Strap	Aluminum Sheet $1\frac{1}{4}$ x 17
13	2	Compression Fitting	$\frac{1}{8}$" NPS
14	1	Nipple	$\frac{1}{4}$ VPS
15	4	Bolt	$\frac{3}{16}$" brass, length $19\frac{1}{4}$"
16	6	Machine Screw	#8 round head, length $\frac{1}{2}$"

TEST SECTION

Thesis Project: The Effect of Vibration on Heat Transfer by Forced Convection in a Horizontal Cylinder

Lt. R.M. George
Lcdr. W.A Grossetto

7 April 1953

U.S Naval Postgraduate School
Monterey, California

Scale 12" = 1'

Figure IX

JUL 2
SEP 10
JAN 22
FEB 23
MAR 22
APR 19

BINDERY
319
RENEWED
RENEWED
RENEWED
RENEWED

20648
Thesis
G86
Grossetta
The effect of vibration on heat transfer by forced convection in a horizontal tube.

BINDERY
319
SEP 10
OCT 1
SEP 14 '53
OCT 13 '53
NOV 20 '53
RENEWED
RENEWED
RENEWED
RENEWED
RENEWED
JAN 22
FEB 23
DISPLAY

Thesis
G86
Grossetta
20648
The effect of vibration on heat transfer by forced convection in a horizontal tube.

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